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by

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ABSTRACT The results of the study of the temperature field of the cooled drum rotor of the SKR-100 turbine presented in this paper complete the investigation of the static thermal model of the rotor. The results describe temperature fields in the case of circular channels distributed in such a manner that the rate of outflow from the preceding channels is fully quenched. It is shown that the circular channels in the SKR-100 turbine under investigation produce the same degree of cooling as the elliptic channels within staggered blade distribution used by KhTGZ im. Kirova. In view of the greater technological simplicity of circular channel production as compared with the oval ones, the new approach is recommended for the authors for use in the future design of supercritical turbines (up to 400 at. abs., 700C). The substitution of circular channels does not seem to affect the nonstationary temperature field. Orig. art. has: 6 formulas, and 1 table. English Translation: 14 pages.

TEMPERATURE FIELD OF A TURBINE ROTOR COOLED BY VAPOR PASSING THROUGH CIRCULAR CHANNELS IN THE BLADE SHAFTS

D. A. Pereverzev, L. V. Povolotskiy and N. B. Chirkin

This paper presents the results of investigations of the stationary and nonstationary temperature fields in a cooled rotor of a steam turbine. The derived data can be used in creating and constructing powerful promising turbogenerator sets for supercritical steam parameters.

The data from investigations of the temperature field of a cooled drum rotor of a SKR-100 turbine presented in this article complete work on a static thermal model of a rotor, some results of which were presented in [1, 2].

In contrast to the preceding investigations [1, 2], the oval cooling channels in the blades and spacers have been replaced by round cooling channels (Figure 1): in the blade at a distance of a half-pitch there are two channels ∅ 7 mm; in the spacer -- one channel in the center ∅ 10 mm. When the channels are made in this way the position of the blades and spacers in the model ensured complete extinction of velocity of outflow from the preceding channels and the maximum possible efficiency of cooling in the transition chamber between adjacent rows of blades and spacers.

The mean cross section of the model, for which the experimental results were analyzed in most cases, and views of the butts of the blade and spacer are shown in Figure 1.

The method for measuring the temperatures of points of the model and the parameters of the heating and cooling steam, the number of measured points and their position remained the same as in the preceding investigations [1, 2].

The results of investigations of the stationary temperature field were analyzed in the form of a dimensionless dependence, using which it then is possible to determine the efficiency of cooling of a real rotor

$$\bar{t} = \frac{t - t_0}{t_h - t_0} = \Phi(\mathrm{Bi}_0). \qquad (1)$$

Here $\bar{t}$ and t are the relative excess temperature and the temperature at the considered

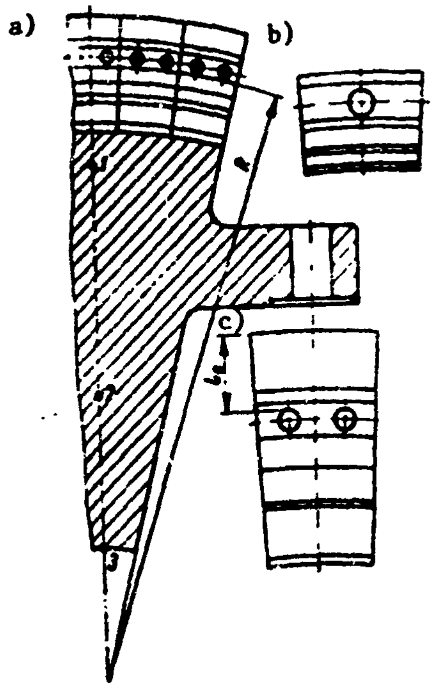

Figure 1. Mean cross section of thermal sector model of rotor (a) and appearance from end of shaft (along path of steam) of spacer (b) and blade (c):

1, 2 and 3 — points for which an analysis of the results of investigations was made for stationary and nonstationary temperature fields.

point of the sector; t_h and t_0 are the mean temperatures of the heating and cooling vapor; $Bi_0 = \alpha_0 l_0/\lambda$ is the Biot criterion in the direction of cooling in the mean cross section of the second row of blades; α_0 is the heat transfer coefficient in the direction of cooling in the channel of the blade; l_0 is the determining dimension in the direction of heat transfer (distance from the heat supply surface to the upper point of the channel of the blade); λ is the coefficient of heat conductivity of the material of the blade.

Equation (1) is not a generalized dimensionless dependence because it does not take into account the heat exchange criteria in the direction of heating and in the transitional chamber between adjacent rows of blades and spacers. However, for this model the assumptions made, as revealed by preceding investigations [1], are completely justified and the three-dimensional problem can be reduced to two-dimensions.

Two groups of regimes were considered in the course of investigating the efficiency of cooling:

1) special baffles inserted in the heating channel [1] were completely open; in addition, at the entrance and exit of the apparatus the maximum possible discharge of heating steam passed along the heating channel at stipulated pressures;

2) the heating steam baffles were closed in such a way that on each of them there was a pressure drop of 1.5-3 kg/cm^2[(1.47-2.94)$\cdot 10^5$)N/m^2].

In order to ensure $Bi_0^m = Bi_0^n$ = 4.5-5 (equality of the Biot criterion in the model and in the real apparatus at the rated operating regime of the turbine) the pressure at the entrance to the investigated cooling channel (second row of blades along the path of the steam) was maintained at ∿ 15 atm (abs.) (1.47 $\cdot 10^6$ N/m^2). In most of the rated regimes the pressures at the entrance to the second row of blades and along the heating channel before the first baffle were maintained identical.

The discharges and the temperature of the heating and cooling steam were maintained the same as in the preceding investigations [1]. The temperature of the steam beyond the cooler was maintained in the range 220-240°C with a safety factor in relation to saturation temperature not less than 10-15°C; the temperature of the heating steam in the apparatus on the average was 300-325°C; the discharges of the cooling vapor varied in the range 800-1600 kg/hour (corresponding to Bi_0 = 3.0-5.5), the velocities in the investigated channel in this case attained 110-170 m/sec and the heat transfer coefficients were 2100-4200 W/m$^2\cdot$degree; the expenditures of heating steam varied in the range 2-5 tons/hour. In addition, the velocities through the heating channel were 300-450 m/sec.

A total of 2.5-3 hours of continuous operation of the apparatus usually is required for attaining a steady-state regime.

In all regimes the model first was heated in such a way that the temperature of the sector was greater than the mean temperature of the cooling vapor by 30-40°, after which cooling steam at a stipulated temperature was fed into the apparatus. As a result of the duration of the process and fluctuations of the steam parameters

in the thermoelectric power station system the temperature field usually was brought only to a near-stationary state. In combination with additional heating through the flanges [3] this caused somewhat exaggerated values $\bar{t}$. Therefore, for determining the temperatures of the metal in the drum of a real rotor we used the relative temperature $\bar{t}_1$ of a point of the mid-section situated closest to the heat transfer surface (point 1 in Figure 1); at all the remaining points of the section the temperatures in the real rotor to all inteuts and purposes were equalized to the temperature $\bar{t}_1$, not exceeding it in value (see [1, 3]).

Since the earlier experiments revealed that the temperature of the surface of the blades is close to the mean temperature of the heating steam, at this stage it was decided to compute the values $\bar{t}$ for the mean temperature of the heating system, assuming the Biot criterion in the direction of heating to be equal to infinity [1].

The values α_0 in the channels of the blades were determined on the basis of the known dimensionless dependence for short pipes

$$Nu = 0{,}021\, Re^{0.8} Pr^{0.43} \varepsilon_l. \qquad (2)$$

In computations using equation (2) the mean temperature of the cooling steam was assumed to be decisive.

For the second group of regimes with identical Bi_0 the values $\bar{t}$ are 0.02-0.03 greater than for the first, which agrees with the data of the preceding experiments [1]. However, since this difference is small, and the region of Bi_0 is wider than

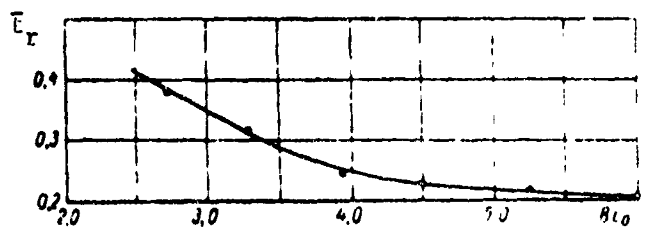

Figure 2. Experimental curve $\bar{t}_r = f(Bi_0)$ for variant with round cooling channels.

covered by the data for the first group, they were used (Figure 2) in computing the temperatures arising in the drum of the rotor of a SKR-100 turbine. Computations

were made for the rated operating regime of the turbine. The values λ (EI612 steel) were determined from the averaged temperature of the blade shafts.

The temperature of the cooling vapor for all the steps was assumed to be 525°C; in addition, it was assumed that the increase of vapor temperature resulting from heating is compensated by its decrease during throttling in the cooling channels.

The results of computations are given in the table, which shows that the maximum temperature of the rotor drum is 547°C; it decreases along the path of the steam. However, the axial drop of temperature is small and for the entire rotor is 11°C. This indicates that the assumption of equalization of temperatures in the axial direction, used as the basis in modeling and analysis of the experimental results, is correct.

Notation and dimension	Number of stage									
	2	3	4	5	6	7	8	9	10	11
t_h in °C	622	614	607	600	592	585	577	569	562	554
t_0 in °C	525	525	525	525	525	525	525	525	525	525
α_0 in W/m^2·degree	5050	4550	4350	4130	4000	3765	2620	3500	3290	3090
Bi_0	4.5	4.11	3.98	3.76	3.65	3.41	3.32	3.23	3.03	2.85
$\bar{t}_r$	0.223	0.238	0.246	0.264	0.274	0.300	0.310	0.320	0.346	0.368
t_r in °C	547	546	545	545	543	543	541	539	538	536

The temperatures were determined without taking into account the heat loss into the zone of the first (active) stage and the front end packing where the rotor is bathed by the cooling vapor over the entire surface. In addition, as shown by an analysis, the throttling predominates over heating and as a result the temperature of the cooling vapor decreases from stage to stage. Therefore, the temperature of the rotor drum scarcely exceeds 547°C, which makes it possible to make it from pearlite steel.

Some of the experiments were devoted to investigations of the nonstationary temperature field. The purpose of these investigations was determination of the temperature differences in the body of the sector in dependence on time, determination of the zone of the regular regime of the model for different heating regimes and the use of the results in estimating and checking the thermal behavior of the rotor under natural conditions.

It should be noted that in comparison with real size in the model there was a disruption of the geometry (presence of flange connections) and additional boundary conditions were involved (supplying of heat through the flanges, heat exchange with the surrounding medium), and as a result there was some disruption of the axially symmetrical character of the problem. As demonstrated by an analysis of the stationary temperature field [1, 3], this leads to an estimate of the temperature level of the cooled rotor with some margin. To some degree this should be expected also in investigations of the nonstationary temperature field. In particular, the temperature differences between the periphery and hollow of the rotor drum, arising under natural conditions, on the basis of experimental data for the model (as a result of the presence of a flange in it) also will be estimated with some margin.

In the course of these tests the pressure at the entrance to the heating and cooling channels in most of the regimes also was maintained at $\sim$ 15 atm (abs.) $1.47 \cdot 10^6$ N/m^2), which ensured an approximate equality of the Biot criteria in the model and the real object for the nominal operating regime.

In order to keep the stationary boundary conditions necessary for investigating the regular regime of the first kind [4], the vapor lines to the enclosure of the model were first heated with the maximum possible expenditure of live steam. This in part corresponds to real conditions because in most cases the lines to the turbine enclosure are heated first and only then does the heating of the turbogenerator begin.

After a stable steam temperature is established the heating of the model was done in stages. The pressure was increased in three equal stages at intervals of

10-15 minutes each. The temperature and pressure of the steam, corresponding to the stipulated regime, usually were established only 30-40 minutes after the onset of heating.

All the regimes with stationary boundary conditions were broken down into three groups, when heating of the model was accomplished: 1) only in the heating channel; 2) only in the cooling channel; and 3) in both channels simultaneously. In addition, in the first and third groups there were regimes with fully raised and partially lowered heating steam baffles.

Some of the experiments were made with a gradual increase (small stages) of pressure of the heating medium with time.

The temperature of the points of the model, blades and spacers were recorded each 4-5-minutes with calibrated Chromel-Kopel thermocouples. The points for which the analysis of the experimental results was made are shown in Figure 1. Figure 3 shows the curves $t = f(\tau)$ for these points for the regime of the third group with lowered baffles at a temperature of the heating vapor of 300°. At all the remaining points of the model the curves have a similar character.

Figure 3 also gives $\ln\theta = \ln(t_{st}-t) = \varphi(\tau)$ which were used in determining the time of onset of the regular regime τ_r and the values of the heating rates m(t_{st} -- stationary temperature of the point). These curves show that a regular regime sets in 1-1.2 hours after onset of heating; for the lower point 3 it is 25-30 minutes later than for the upper point 1.

In the experiments it usually was not possible to attain the values of the stationary temperature of the points of the sector due both to the duration of the process of heating and due to fluctuations of steam vapor. Therefore, the value t_{st} was determined using the formula from [2]

$$t_{cT} = \frac{t_2^2 - t_1 t_3}{2t_2 - (t_1 + t_3)}, \quad (3)$$

Legend: CT = st; where t_1, t_2, t_3 are the temperatures of the point in sequence of increase, separated by equal time intervals $\Delta\tau$ on the segment of the temperature

curve situated presumably in the zone of the regular regime.

The value t_{st} then was averaged for values differing from one another by not more than

The values of the heating rates m for points 1 and 3 were determined from the $\ln\theta = \varphi(\tau)$ curves as the tangents of the angle of slope of the linear segments and were computed using a formula from [2]

$$m = \frac{1}{\Delta\tau} \ln \frac{t_2 - t_1}{t_3 - t_2}, \tag{4}$$

where t_1, t_2, t_3 are similar to relation (3), but the time interval $2\Delta\tau$ covers the entire zone of the regular regime. The values m, determined by these methods, coincided well with one another, which made it possible to determine the values of the heating rate for all the remaining points of the model using formula (4); in addition, in order to guarantee the choice of the temperatures t_1, t_2, t_3 in the zone of the regular regime it was assumed $\tau r = \tau r_3$ (Figure 3), since point 3 enters into the regular regime after the others.

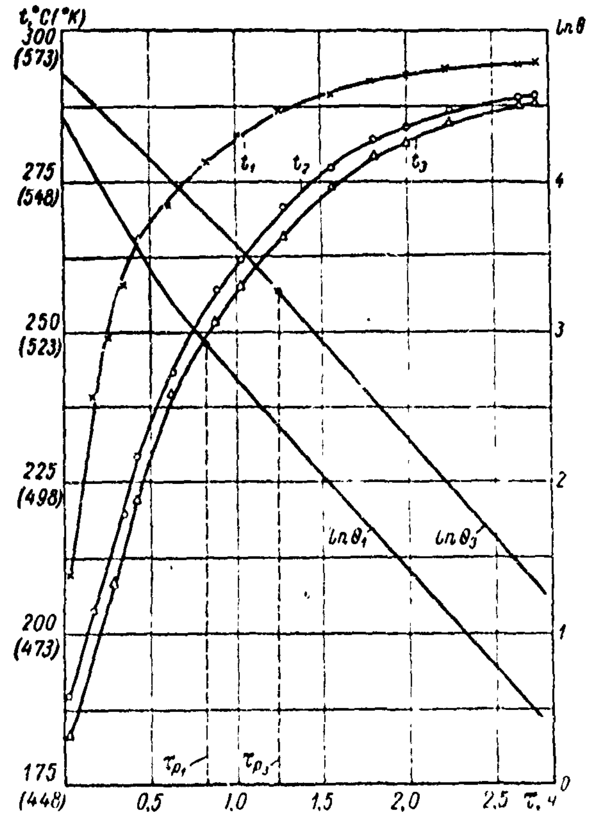

Figure 3. Curves $t = f(\tau)$ and $\ln\theta = \varphi(\tau)$ for points of the mean cross section of the sector model during heating by system t = const for both channels simultaneously. 4 = hours.

Computations revealed that for each individual regime at all points of the sector the values m are virtually equal to one another. For different regimes the value of the heating rate varies in the following limits for the 1st, 2nd and 3rd groups respectively: 1.26-1.34; 1.28 and 1.27-1.32 1/hour. In addition, in the limits of the group the larger values m apply to the case of partially lowered heating steam baffles, when there is an increase of the intensity of heat exchange in the direction of heating.

In order to estimate the reliability of the data obtained using the model the value of the rate of heating in the real rotor was estimated by computations using the relation

$$m = \left(\frac{\beta_1^2}{\delta^2} + \frac{\beta_2^2}{R^2}\right) a, \tag{5}$$

where β_1 and β_2 are the first roots of the transcendental equations for a cylinder of finite length, placed in a heating medium with the boundary conditions of the third kind; 2δ, R and a are the length, peripheral radius and coefficient of thermal conductivity of the rotor drum. Here the following assumptions were made: the end parts of the rotor were disregarded; the rotor drum is considered as a cylinder of finite length without a central opening; the rotor drum is bathed by the heating medium with a temperature identical on all surfaces.

A mean rotor drum temperature of 540°C (rated operating regime) was used in estimating the value a. We considered two values of the heat transfer coefficient α along the cylindrical surface of the drum: 580 and 2320 W/m^2·degree, used in determining the Biot criterion, and the tables in [5] were used in finding the roots β_1 and β_2; in both cases the heat transfer coefficients at the ends of the drum were assumed to be 2320 W/m^2·degree. As a result of this computation m = 1.21 and 1.4. A further increase of α up to infinity to all intents and purposes has no effect on the value m. To some degree this justifies the stability of the values m obtained in the thermal model for all regimes in this stage of the investigations

since the heat transfer coefficients, reduced to the peripheral radius R of the sector model, fits entirely in the interval 580–2320 $W/m^2\cdot$degree.

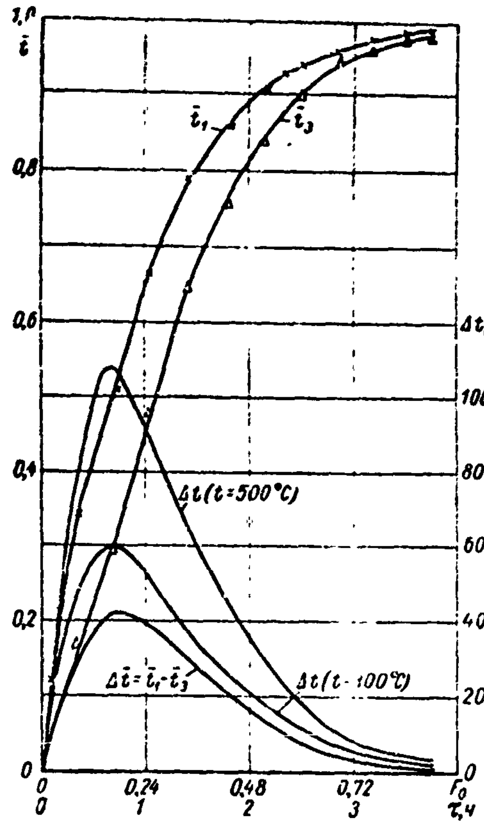

Figure 4. Curves of the relative temperatures $\bar{t} = f(F_0)$ and the temperature differences $\Delta t = t_1 - t_3 = f(F_0)$ of points 1 and 3 of the rotor drum with heating by vapor t = const for both channels simultaneously; $\Delta t = t_1 - t_3 = f(\tau)$ are the curves of the temperature differences at these same points of the drum of a real rotor with the heating of the latter by steam, t_h = 300 and 500°C for both channels simultaneously. 4 = hours.

For determining the temperature differences arising during heating under natural conditions the temperature curves of the points of the model were represented in relative units

$$\bar{t} = \frac{t-t_{in}}{t_{st}-t_{in}} = f(F_0), \qquad (6)$$

where $\bar{t}$, t and t_{st} are the relative, current and stationary temperatures of a point of the model; t_{in} is the initial temperature of the model; $F_0 = a\tau/R^2$ is the Fourier criterion; R is the determining dimension of the model (peripheral radius of the

rotor drum); τ is time.

In investigations and analysis of the experimental results preference was given to heating regimes only along the heating channel or along both channels of the model simultaneously, because in the start-up adjustment tests of the turbine these regimes apparently will be determining.

Figure 4 shows the curves $\bar{t}$ and $\Delta\bar{t} = f(F_0)$ for the heating regime of the model simultaneously in both channels and revealed changes of the temperature difference between points of the rotor drum of the turbine corresponding to points 1 and 3 in the model during heating of the turbine from a cold state ($t_{in} = 25°C$) by steam with a temperature of 300 and 500°C (in the computations it was assumed that $t_{st} = t_h$). For $t_h = 300°C$ the maximum temperature difference between the periphery and hollow of the drum is ∿ 60°C, for $t_h = 500°C$ -- up to 110°C. These values virtually coincide with the data obtained in a similar regime for the variant with oval channels [2].

The stationary temperature field in both cases is established ∿ 3 hours after the onset of heating (taking into account the lag in a triple-step increase of steam pressure). The lag in the step-by-step increase of pressure (temperature) in a real rotor must not be less than in the model. In order to decrease the temperature difference in the second case it is desirable to bring the temperature field of the rotor to a nearly stationary state at a steam temperature of 300°C, when the difference is not more than 60°; then the steam temperature is raised in steps (with a total lag of 30-40 minutes) to 500°. The temperature difference along the radius of the drum in this case will not exceed 60°. However, the total heating time increases to 5-6 hours.

Figure 5 gives the temperature curves of points 1 and 3 of the sector during heating along both channels by steam whose temperature increases continuously in conformity to the law $t_h = f(\tau)$. This pattern was governed by the graph of increase of pressure p_h in the apparatus in conformity to a linear dependence. The temperature difference Δt between the points 1 and 3 in this case does not exceed 60°C.

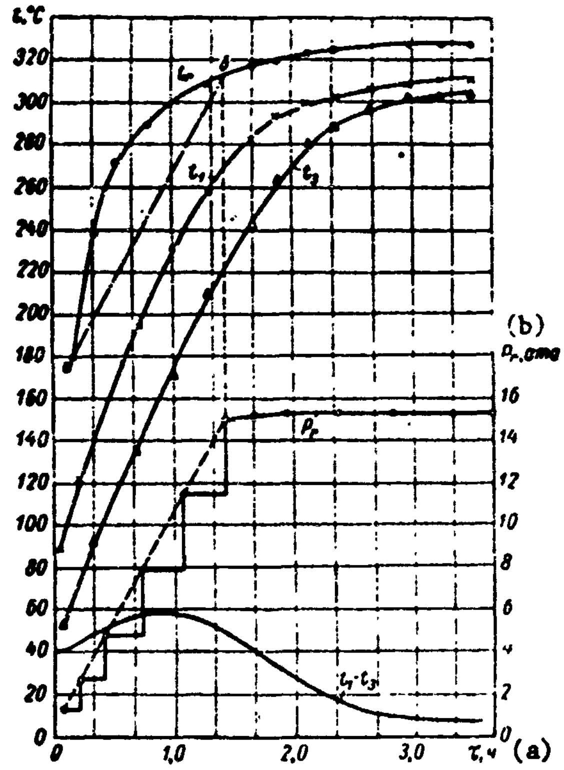

Figure 5. Curves of temperature t = $_4(\tau)$ and temperature differences Δt = f(τ) of points 1 and 3 of the rotor drum during heating of the latter along both channels simultaneously by steam whose parameters vary with time. Legend: a = hours; b = atm (abs.).

In this regime the maximum value of the **steam** temperature (320°C) was attained ∿ 2 hours after the onset of heating. Figure 5 shows that in the case of an increase of vapor temperature t_h in conformity to the linear law (dash-dot segment aσ) the temperature change Δt should decrease, which makes it possible to decrease the time of heating of the rotor. Taking into account the fact that the boundary conditions in the nonstationary regime for the model and the real object are the same, it can be concluded that it is desirable, from the point of view of decreasing the time of heating with admissible temperature differences on the rotor, to have a continuous increase of the steam temperature in conformity to a linear dependence. The approximate computations reveal that the time for heating of the rotor (with an increase of t_h to 500°C) can be shortened by 2-2.5 hours in comparison with the stepped increase of steam temperature considered above (to 300 and 500°); the maximum value Δt in the case of a linear increase of steam temperature will be even some-

what lower.

On the basis of the above the following conclusions can be drawn:

1 In the case of equal expenditures of cooling steam the system for cooling of the SKR-100 turbine rotor with round channels in the blades and spacers for all intents and purposes ensures the same efficiency of cooling as the variant of a checkerboard arrangement of blades and spacers with oval cooling channels which was adopted by the Khar'kov Turbogenerator Plant imeni S. M. Kirov as final [1]. Since round channels are easier to produce than oval channels, the considered variant can be recommended for use in creating powerful turbines for supercritical steam parameters (to 400 atm (abs.), 700°C).

2. The replacement of oval openings by round ones exerts no influence on the heating regimes of the rotor (nonstationary temperature field). The values of the rates of heating and the temperature differences between the periphery and the hollow of the rotor drum remained at the same level as for the variant with oval channels [2]. However, in this case we obtained more stable values of the heating rates: in all the regimes the value m varies about the value 1.3. In addition, it was established that for the peripheral points of the rotor drum the regular regime sets in 25-30 minutes earlier than for points on the surface of the hollow.

3. The experimental data on the heating regimes can be used in an analysis of the optimal regimes of start-up and loading of a SKR-100 turbogenerator and also for evaluating and checking the thermal behavior of a real rotor at the time of turbine tests.

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